und Rohren," Elektrowärme Int., 24, 415-420 (1966).
4. V. V. Gil', I. S. Desyukevich, N. V. Drozdov, et al., in: Heat and Mass Transfer of Cryogenic Fluids in Porous Heat Exchangers [in Russian], Inst. Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR, Minsk (1974), pp. 34-41.

INVESTIGATION OF A GAS REGULATABLE HEAT PIPE WITH A WICKED RESERVOIR

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Results are presented of an analytical and experimental investigation of the characteristics of a heat pipe with a wicked reservoir as the temperature of the cooling medium, and the heat supply intensity change.

Gas regulatable heat pipes (GRHP) with a wicked reservoir are used successfully to stabilize the temperature of heat-liberating objects operating under substantial changes in the thermal load (10-15-fold) and the cooling temperature, at different temperature levels [1-5]. The extensive application of GRHP of this kind in real apparatus is due to the high stability of the working characteristics, the absence of extremal situations during start-up (e.g., a substantial rise in the vapor pressure and temperature above their working level, as can be observed in structures with a wick-free reservoir when the working fluid is incident) as well as the comparative simplicity of fabrication.

Gas regulatable heat pipes with a wicked reservoir are also the basis for constructing thermostat systems with variable thermal resistance heat pipes which have a higher accuracy of temperature stabilization (e.g., GRHP with electrical feedback, thermodiodes, etc. [2-4, 6]).

The strict dependence of the partial vapor pressure of the working fluid on the reservoir temperature, which governs the circuit operation in many cases, is characteristic for a construction with a wicked reservoir. Using the model of a flat interface between the vapor and the uncondensed gas (UCG) as basis, we obtain a dependence to determine the accuracy of vapor temperature stabilization t_{v_1}, t_{v_2} on the primary parameters t_{c_1}, t_{c_2}, t_{r_1}, t_{r_2}, $V_r/\Delta V_g$:

$$\left(\frac{p_{v2}}{t_{r_2}+273}-\frac{p_{v1}}{t_{r1}+273}\right)-\left(\frac{p_{r2}}{t_{r2}+273}-\frac{p_{r1}}{t_{r1}+273}\right)=\frac{p_{v1}-p_{c1}}{t_{c1}+273}\frac{\Delta V_g}{V_r}, \tag{1}$$

where the partial vapor pressure of the working fluid p is related to the corresponding temperature of the vapor or the vapor—gas mixture t on the saturated curve. Equation (1) is based on conservation of the balance of the mass of gas for two positions of the vapor—gas interface. Its solution for the magnitudes of the pressures $p_{v2}(T_{v2})$ and $p_{v1}(T_{v1})$ permits determination of the accuracy of vapor temperature stabilization $\Delta t_{reg} = t_{v2} - t_{v1}$, and therefore, of the heat liberation source for given ratios $V_r/\Delta V_g$ and temperature fluctuations of the cooling medium $(t_{c1} - t_{c2})$ and of the storage reservoir $(t_{r1} - t_{r2})$. Balance equations of the form

$$Q = k_l (L_к - L_g)(t_v - \bar{t}_c) \tag{2}$$

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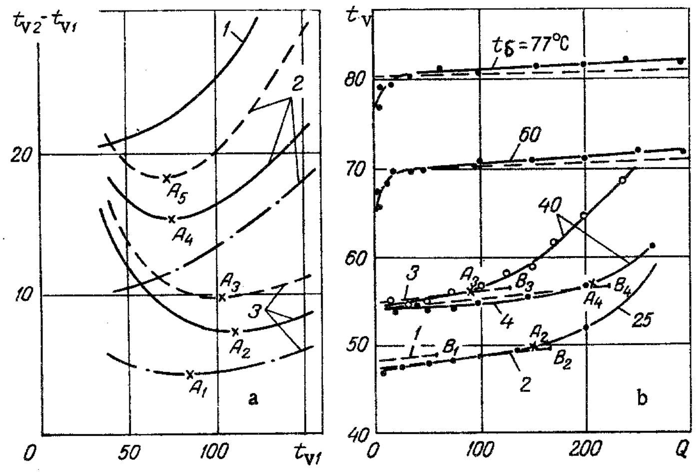

Fig. 1. Stationary characteristics of a GRHP with a wicked reservoir: a) change in thermostat accuracy as the operating temperature level rises (1) $V_r/\Delta V_g = 1$; 2) 5; 3) 10; solid curves, water, t_c = 5-40°C; dashes, methanol; dash—dot 5-20°C); b) characteristics $t_v = f(Q, t_c, t_r)$ (1, 2 — computation for t_r = 25°C and t_c = 40 and 25°C; 3, 4 — computation for t_r = 40°C and t_c = 40 and 25°C. B is the boundary point at which the vapor—gas front approaches the entrance to the reservoir. Points are experimental (open circles t_c = 40°C, dark circles t_c = 25°C, dashed curve, computation), t, °C, and Q, W.

must be explained to take account of the change in heat flux delivered ($Q_1 - Q_2$). However, this requires knowledge of specific values of Q and k_l and complicates the analysis of the operating capacity of the GRHP. In contrast to this, (1) possesses sufficient universality, i.e., it can be used under any heat elimination conditions in the condenser zone.

Results of a computation using (1) under the condition $t_r = t_c$ are presented in Fig. 1a for methanol and water used as working fluids and for two cooling conditions (t_c = 5-40°C for solid curves, and t_c = 5-20°C for the dash—dot curves) and different ratios $V_r/\Delta V_g$($V_r/\Delta V_g$ =1, 5,10). A rise in the working level of the temperature results in a diminution in Δt_{reg} in some temperature range $t_{v1} \leq t_{v1}^{opt}$ (see points A_1-A_5). Therefore, t_{v1}^{opt} determines the working temperature level at which the GRHP with a wicked reservoir has the greatest accuracy of vapor temperature stabilization under given cooling conditions and given $V_r/\Delta V_g$. For water and methanol, the quantity t_{v1}^{opt} lies within the limits 75-115°C for $V_r/\Delta V_g$ = 5-10 and t_c = 5-40°C). The section of the drop in the function $\Delta t_{reg} = f(t_{v1})$ is due to attenuation of the influence of the partial vapor pressure in the reservoir, and the rise section is due to an essential increase in the total pressure, and therefore, the large changes required for advancement of the gas plug. The quantity t_{v1}^{opt} increases with the rise in $V_r/\Delta V_g$ (the points A_2 and A_4) and diminishes with the reduction in t_{c2} (A_1 and A_2). The quantity Δt_{reg} depends on the kind of working fluid and diminishes with an increase in $V_r/\Delta V_g$ (curves 1-3). The essential difference between the curves $\Delta t_{reg} = f(t_{v1})$ for water and methanol is explained by the influence of the physicochemical parameters of the heat carrier, which can be taken into account by using the gas checking coefficient $k_{gc} = r\mu/R_\mu$ [7, 8]. The influence of the coefficient k_{gc} is manifested indirectly in different kinds of the curves p = f(t) for water and methanol, which results in large values of Δt_{reg} for methanol in the long run.

The limitation of the reservoir temperature fluctuations increases the accuracy of sustaining the vapor temperature substantially. If t_r = const, (1) takes the form

$$\frac{V_r}{\Delta V_g} = \frac{p_{v1} - p_{c1}}{p_{v2} - p_{v1}} \frac{t_r + 273}{t_{c1} + 273}. \tag{3}$$

Assumptions of a plane vapor—gas front model, which have been analyzed well enough in [7, 9, 10], are made in deriving (1) and (3). Equation (3) is valid under the conditions T_{v1} >

$T_r \geqslant T_{c2}$ since there will hence always be gas in the reservoir ($p_V > p_r$) and the formation of a second vapor—gas front is eliminated. Equation (3) shows that for given t_{V1} and t_{V2} the ratio $V_r/\Delta V_g$ will be less for low values of t_r (however, $t_r > t_{c1}$). This is achieved by using a different kind of cooling plant. If the condition $t_r \geqslant t_{c2}$ is selected, then heating elements are used, which simplifies the construction. For a given ratio $V_r/\Delta V_g$ the diminution in t_r increases the stabilization accuracy. An increase in t_r permits changing the operating temperature level of the diagram since it causes a rise in pressure in the construction. This principle can be used for thermal stabilization of units requiring a variable stabilization temperature.

The quantities p_V in (1) and (2) can be expressed also in terms of the mass of the UCG, V_r and ΔV_g, which complicates the analysis. The use of p_V, which are directly related to t_V, hence permits operating directly at the given temperature level and not the mass of gas which is a constructive (secondary) quantity.

The thermostabilizing capacities of a GRHP are improved as compared to the stationary operating conditions for a periodic change in the power [11], as well as of the cooling temperature t_c. In this case, the reservoir temperature cannot reach the extreme values $t_{r1} = t_{c1}$, $t_{r2} = t_{c2}$ for a pulse change in t_c in the range t_{c2}-t_{c1}-t_{c2}, etc. but fluctuates in a narrower band between the curves $t_{r1} = f(\tau)$ and $t_{r2} = f(\tau)$ defined by the equations

$$t_{r_n} = (t_{r_{n-1}} - t_c)\left\{0.5\left[\exp\left(-\frac{\tau_h}{\tau_h^x}\right) - \exp\left(-\frac{\tau_c}{\tau_c^x}\right)\right][1 + (-1)^{n+1}] + \exp\left(-\frac{\tau_c}{\tau_c^x}\right)\right\} + t_c; \quad (4)$$

$$t_c = 0.5\,(t_{c2} - t_{c1})\,[1 + (-1)^{n+1}] + t_{c1},$$

where n is the order number of the cooling temperature fluctuations. The points $n = 2i$ form a curve t_{r2}, and the points $n = 2i - 1$ a curve t_{r1}. The computation starts with the initial condition $t_{rc} = t_{ini}$. The reservoir time constants t_h^x and t_c^x ($\tau^x = (mc)R$, where R is the thermal resistance between the reservoir and the environment) depend on the cooling intensity, the type of heat insulation, and the construction of the reservoir and condenser junctures, the specific heat of the reservoir, and can be estimated as follows:

$$\tau^x = (mc)_r\left\{\pi L_T \Big/ \left[1/\alpha d_r + \sum_{i=1}^{k} (1/2\lambda_i)\ln(d_{i+1}/d_i)\right] + \left[L_T/\lambda_T F_T + 1/f_K\lambda_K F_K \operatorname{th}(f_K L_K)\right]^{-1}\right\}^{-1}; \quad (5)$$

$$f_K = (\pi\alpha d_K)^{1/2}\cdot(\lambda_K F_K)^{-1/2},$$

where λ_i and d_i are the heat conduction and diameter of the i-th layer of heat insulation. Applying the appropriate type of heat insulation (selecting λ_i and d_i) and diminishing the heat transfer along the heat-pipe housing (diminishing λ_T, F_T) we can reach very large values of τ^x, which permits diminution of the reservoir temperature fluctuations and raising the accuracy of diagram operation for a periodic change in temperature of the environment.

Nonstationary GRHP modes are discussed insufficiently completely in the literature although they are of scientific and practical interest. The GRHP should function under conditions when many perturbing parameters vary with time. A model of nonstationary GRHP modes is presented in this paper, which takes account of the possibility of a change in Q, T_c, α_{hk} in time, and permits determination of the function $T_V = f(\tau)$ for a given heat pipe geometry, and therefore, of the time of GRHP arrival at the mode. This target is indeed due to another approach to the compilation of the equations for nonstationary modes and expressions (1) and (3).

A system of nonlinear heat-balance differential equations for the evaporator, the transport zone, and condenser was compiled to compute the nonstationary modes of heat-pipe operation. The following assumptions were made: a) the vapor—gas front has a sharp interface; b) heat outflows from the evaporator up to the arrival of the front from this zone are due to heat conduction along the housing walls; c) heat transfer by the vapor flux is realized after the vapor temperature t_V' at which there is no UCG in the evaporator, is reached; d) the temperature drop between the evaporator wall and the vapor during GRHP operation does not introduce substantial error in the computation of the transient mode duration.

Cleansing of the evaporator of gas, as well as heating it as characterized by the heat flux

$$Q = [(mc)_{hs} + 0.5\,(mc)_{tr}]\, dt_v/d\tau + rV_{hs}\, d\rho_v/d\tau + (t_v - t_c)/\{L_{tr}/\lambda_{tr} F_{tr} + [\lambda_k F_k f_k \,\mathrm{th}\,(f_k L_k)]^{-1}\} \quad (6)$$

occurs in the vapor temperature range $t_c < t_v < t_v'$. The front advances into the transport zone when the vapor temperature t_v' is reached. The heat flux goes into heating the evaporator zone, the transport zone, and the outflows:

$$Q = \frac{dt_v}{d\tau}\{(mc)_{hs} + (mc)_{tr}\,[1 + (1 - \nu)\, L_k/L_{tr}]\} + (t_v - t_c)\left[\frac{(\nu - 1)\, L_k}{\lambda_{tr} F_{tr}} + \frac{1}{\lambda_k F_k f_k \,\mathrm{th}\,(f_k L_k)}\right]^{-1}. \quad (7)$$

When the front enters the condensation zone $t_v > t_v''$, the following equation is used

$$Q = \frac{dt_v}{d\tau}[(mc)_{hs} + (mc)_{tr} + \varepsilon_k (mc)_k (1 - \nu)] + (t_v - t_c)\, k_l L_k (1 - \nu), \quad (8)$$

where $\nu = L_g/L_k$ and is presented in [12], and ε_k takes account of the difference between the condenser and the vapor temperatures. This system can be used in the analysis of not only the starting modes, but also the transients. In this case the appropriate time functions Q, t_c, t_s are substituted into (6)-(8). If the heat flux is delivered to the evaporator from a source, the system of equations is supplemented by a heat balance equation for the source

$$Q = (mc)_s \frac{dt_s}{d\tau} + \frac{t_s - t_v}{R_s} + \frac{t_s - t_{env}}{R_{env}}, \quad (9)$$

where t_s is the temperature of the source and R_s, R_{env} are the thermal resistances between the source and the evaporator, and the source and the environment. The quantity $(t_s - t_v)/R_s$ is hence substituted into (6)-(8) instead of Q. The solution of system (6)-(9) is not difficult when using numerical methods, and can be performed on small electronic computers.

The assumptions expressed about the influence of the cooling temperature, the working temperature level, and the reservoir temperature on the vapor temperature, about the possibility of raising the accuracy of regulation for a periodic change in the cooling temperature were confirmed on a heat pipe (HP) with a wicked reservoir with the following geometric characteristics: total length L_{HP} = 620 mm, reservoir length L_r = 170 mm, HP diameter d_{HP} = 13 mm, and reservoir diameter d_r = 26 mm. The wick was fabricated from sintered copper matting, has a porosity Π = 92% and thickness of 2 mm in the heat supply and removal zones, and 1 mm in the reservoir. The transition of the condenser wick was accomplished by overlapping and scorching it to the reservoir wick. The working fluid used was water, and the UCG was nitrogen. Experiments were conducted on the test stand described in [12].

The stationary GRHP characteristics GRHP $t_v = f(Q, t_c, t_s)$ are shown in Fig. 1b. The GRHP was investigated at the temperature levels t_v = 55 and 80°C by a change in the UCG mass at a 20-25 and 40°C cooling temperature and a heat load in the 5-300 W range.

The regulation started with 5-10 W because of the high thermal resistance of the transport zone in the axial direction, which was achieved by diminishing the housing wall thickness. As the load increased, the front was displaced from the transport zone into the condenser and then approached the entrance to the reservoir. As the load increased further, the vapor—UCG boundary leaves the condenser zone and exerts influence on the reservoir temperature mode by increasing its temperature, and therefore, changing the slope of the curve $t_v = f(Q, t_c)$ (the points A_2-A_4).

As the cooling temperature rises from 25-40°C, a rise in the working temperature level occurs, caused by the increase in the partial vapor pressure of the working fluid. The experiment showed that the temperature rise depends on its nominal value. Thus, the rise was ∿ 7°C at the 50°C level, and ∿ 4°C at the 80°C level.

The construction of the reservoir in the diagram under investigation permitted varying its temperature by using a heater mounted in the reservoir. An increase in the reservoir temperature causes a rise in the characteristic $t_v = f(Q, t_c)$ with a practically nonessential change in steepness. The maximum heat flux transmitted is hence increased (see Fig. 1b). The influence of the reservoir temperature on the temperature level grows with the rise in its temperature. Thus, for t_r = 30°C, the quantity dt_v/dt_s is ≈ 0.275°C/°C, but 0.5°C/°C at the 40°C level, and 0.7°C/°C at the 70°C level.

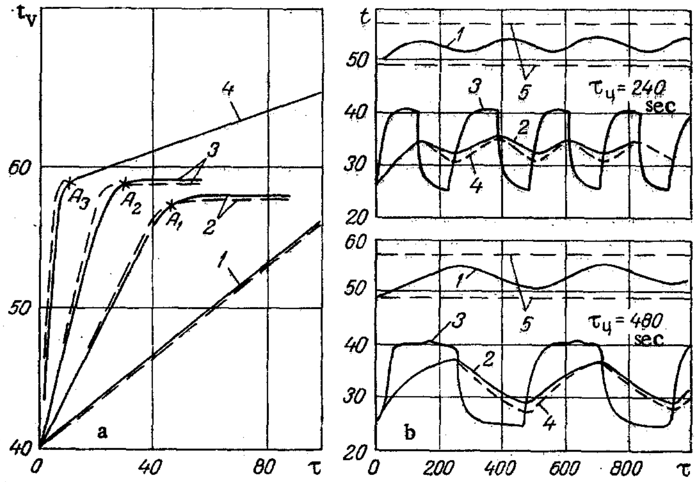

Fig. 2. Nonstationary GRHP characteristics: a) start-up at an initial temperature t_c (1 — Q = 10 W; 2 — 25; 3 — 50; 4 — 200; solid curves, experiment; dashes, computation; t_c = 40°C); b) functioning of the construction at a variable cooling temperature (1, 2, 3 — temperatures of the vapor, reservoir, and of cooling; 4 — computation; 5 — limits for stationary conditions; Q = 100 W). t, °C, τ, sec.

Investigations performed without a load on the heat pipe permitted a deduction that the operation of the reservoir is subject to the fundamental regularities of operation of the GRHP supply zone: for small heat loads the thermal flux is removed by heat conduction and diffusion, and the vapor—gas mixture is inside. As the heat flux increases, the gas concentration drops and the supply zone is cleansed of UCG and the interface emerges in the condenser domain.

The starting characteristics of a heat pipe are displayed in Fig. 2a. The curves $t_v = f(\tau)$ can be separated provisionally into two domains distinguished by the tempo of temperature growth (approximate boundary of the points A_1-A_3). The first period (to the point A_1), in which the evaporator is cleansed of the UCG which emerges in the transport zone (≈ 80-95% of the time), is the longest. As the starting power grows, the start-up duration diminishes. For a 200-W load (curve 4), which exceeds the maximal heat exchange in the possibilities in the condenser, an additional rise occurs in the temperature because of heating of the reservoir (section after A_3).

Certain results of GRHP testing under cyclically varying cooling temperatures (range of variation 25-40°C, 4-8 min period) are shown in Fig. 2b. The change in temperature of the vapor (curve 1), the reservoir (curve 2), and the cooling medium (curve 3) is shown in the graphs, and the boundaries (5) corresponding to the stationary GRHP characteristics are presented also. The change in reservoir temperature occurs in a narrower band than the cooling temperature, and its fluctuations diminish with the rise in pulse repetition rate. The quasistationary mode sets in right after the first cycle. Such partial temperature stabilization of the reservoir raises the accuracy of vapor temperature stabilization. Thus, for an 8-min period (4 min heating and 4 min cooling), the mean reservoir temperature was in the 30-36°C band while the vapor temperature corresponded to 53 ± 2°C as compared with ± 3.5°C for stationary conditions.

Connecting the regulating unit to the reservoir heater permits an improvement in its thermal stabilization and attainment of independence of its temperature from the nature of the change in environment. Selected as reservoir reference temperature was t_{c2} = 40°C, which diminishes the energy expenditure. The power delivered to the reservoir heater should compensate for the losses at the greatest temperature drop between the reservoir and the environment

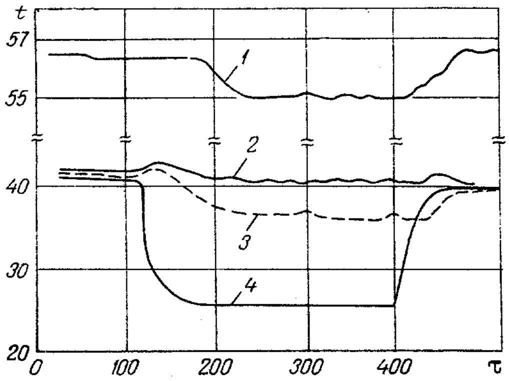

Fig. 3. Operation of a GRHP with a thermostatted reservoir for a variable cooling temperature: 1, 2, 3, 4) temperatures of the vapor, reservoir, points at the endfaces, cooling medium, t, °C; τ, sec.

$$Q_r = \frac{\pi L_r (t_{c2} - t_{c1})}{\frac{1}{\alpha d_r} + \sum_{i=1}^{k} \frac{1}{2\lambda_i} \ln \frac{d_{i+1}}{d_i}} + \frac{t_{c2} - t_{c1}}{\frac{L_T}{\lambda_T F_T} + \frac{1}{\lambda_K F_K f_K \operatorname{th}(f_K L_K)}} + Q_d, \tag{10}$$

where Q_d is defined by [9].

The stationary GRHP characteristics are shown in Fig. 1b for t_r = 40°C. A change in cooling temperature from 25 to 40°C causes a 1.5°C rise in the vapor temperature level, which is substantially better than for a GRHP with a wicked reservoir for $t_r = t_c$ = var.

Dependences of the change in vapor temperature (curve 1) during operation of a GRHP with a wicked reservoir are shown in Fig. 3. Thermal stabilization of the reservoir was accomplished by using a two-position regulator. The operating mode with Q = const and t_c = 40 → 25 → 40°C was examined. A diminution in t_c results in the appearance of heat outflows in the condenser along the GRHP housing, which is related to disturbance of the isothermy of the reservoir (compare curves 2-3, corresponding to temperatures along the reservoir edges). This phenomenon is weakened substantially upon applying a high thermal resistance zone connecting the reservoir and the condenser [2, 6]. Fluctuations in the vapor temperature with a 15°C change in cooling temperature conditions were 1.5°C, which is in good agreement with the results of operation of the unit in the stationary state.

The results of experimental investigations of the stationary and transient characteristics were compared with computed models. Theoretical characteristics for the stationary conditions are shown in Fig. 1b. The greatest difference as compared with experiment is noted in the range Q < 5-10 W, which is not a GRHP working range. The maximum heat flux at which the front approaches the entrance to the reservoir, but still does not act on its temperature mode is correctly estimated by the computation (compare A_2-A_4 with B_2-B_4). Sufficiently good agreement between theoretical results and experiment is visibly explained by the fact that the blurred vapor—gas front does not exert substantial influence on the integrated characteristics $t_v = f(Q, t_c)$ under given cooling conditions and the constructive heat pipe parameters since the mass of gas contained therein is small compared to the total mass of UCG in the system. Heat conduction along the housing wall also does not introduce substantial errors for this case. The heat flux transferred over the housing into the vapor—UCG zone is ≈ 3-5% of the flux computed by the plane front model. Computations using (6)-(8) for the starting modes exhibited a nature of the change in vapor temperature with time similar to the actual (see Fig. 2a) and permit determination of the duration of GRHP start-up. The existing differences are probably associated with the influence of heating of the heat supply zone (3-5°C) and the spiral of the ohmic heat pipe heater, the inaccurate determination of overflows along the housing, and also diffusion processes. Calculations using (4) and (5) afford a possibility of determining the nature of the change in reservoir temperature (Fig. 2b), and therefore, of estimating the vapor temperature fluctuations, and the increase in thermostat accuracy for a periodic change in cooling temperature.

NOTATION

Q, heat flux; t, temperature; p, ρ, pressure and density of saturated vapors of the working fluid; V, volume; τ, time; τ^x, reservoir time constant; mc, total specific heat; λ, heat-conduction coefficient; L, length; d, diameter; F, cross-sectional area; k_l, heat-transfer coefficient from the vapor to the cooling fluid referred to unit length of the condenser.

Subscripts: hs, heat supply zone; tr, transport zone; k, condenser zone; T, zone connecting reservoir and condenser; v, vapor; g, gas; c, cooling medium, cooling; d, heat transmission by diffusion; h, heating; 1, 2, minimum and maximum values of Q and t_c.

LITERATURE CITED

1. J. D. Hinderman, E. D. Waters, and R. V. Kaser, "Design and performance of noncondensible gas-controlled heat pipes," AIAA Paper, 420 (1971).
2. F. Edelstein and R. J. Hembach, "Design, fabrication and testing of a variable conductance heat pipe for equipment control," AIAA Paper, 422 (1971).
3. J. P. Kirkpatrick and B. D. Marcus, "A variable conductance heat pipe. Radiator for lunar surface magnetometer," AIAA Paper, 27 (1972).
4. M. Groll, M. Hage, W. D. Münzel, and P. Zimmerman, "An electrical feedback controlled high-capability variable conductance heat pipe for satellite application," Proc. First Int. Heat Pipe Conf., Stuttgart (1973).
5. F. Edelstein, "Transverse header heat pipe," AIAA Paper, 656 (1975).
6. J. P. Kirkpatrick, "Variable conductance heat pipe for satellite application," Proc. First Int. Heat Pipe Conf., Stuttgart (1973).
7. S. V. Konev, "Investigation of heat and mass transfer in gas regulatable heat pipes," Author's Abstract of Candidate's Dissertation, Inst. of Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR, Minsk (1976).
8. S. V. Konev, "Heat exchange in gas regulatable heat pipes," in: Intensification of Energy and Material Transfer Processes in Porous Media at Low Temperatures, Minsk (1975), pp. 3-22.
9. P. A. Mirzoyan and O. M. Shalya, "Influence of tube wall heat conductivity and vapor diffusion on the longitudinal temperature profile of a GRHP," in: Tr. Mosk. Energ. Inst., No. 198, 99-104 (1974).
10. Binert, "Application of heat pipes for temperature regulation," in: Heat Pipes [Russian translation], Mir, Moscow (1972), pp. 349-370.
11. L. L. Vasil'ev, S. L. Vaaz, V. G. Kiselev, S. V. Konev, and L. P. Grakovich, Low-Temperature Heat Pipes [in Russian], Nauka i Tekhnika, Minsk (1976).
12. M. G. Semena, V. M. Barturkin, and B. M. Rassamakin, "Investigation of stationary and starting characteristics of a heat pipe with an unwicked reservoir," Inzh. Fiz. Zh., 33, No. 3 (1977).